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The effect of fin-and-tube heat exchanger resistance on inlet flow maldistribution related pressure drop penalties

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Abstract

Transient CFD modelling of two and three row circular fin-and-tube heat exchanger geometries experiencing inlet flow maldistribution are presented. The common separated jet flow regime of industrial air heater inlet headers is approximated through a sudden expansion inlet. The higher resistance of the three row heat exchanger results in a greater spreading of the inlet flow maldistribution when compared to the two row case. One third of the heat exchanger in the two row case receives low flow due insufficient spreading of the inlet flow maldistribution. The lower resistance heat exchanger has on average a 56 percent higher pressure drop penalty over comparative uniform flow cases when compared to the higher resistance three row cases. The results highlight the importance of heat exchanger hydraulic resistance in determining the magnitude of inlet flow maldistribution effects on heat exchanger performance. These results are especially applicable to recuperator, pre-heater and heat recovery installations where commonly a limited number of tube passes are employed.

Introduction

Air heater systems find widespread use in a range of industrial processes such as power generation, spray drying, and pulp and paper production. They provide heated air to process units for heating and/or drying applications. The typical forced draft air heater unit configuration is shown in Figure 1 with the main components being the centrifugal fan and the fin-and-tube heat exchanger. The main components are connected to each other and the downstream process through various ducting components. The ducting also forms the air side headers into and out of the fin-and-tube heat exchanger. Air flow maldistribution in air heater fin-and-tube heat exchangers is common due to the large inlet header expansion between the fan exit duct and heat exchanger inlet face.

Typical inlet header expansion angles (2θ) of between 30° and 90° are used to provide a compact expansion between the fan outlet duct and much larger heat exchanger inlet face. The area ratio increase is typically between 6 and 12 to allow a reduction in air velocity prior to entering the heat exchanger. The larger expansion angles reduce diffuser performance. This involves a combination of lower pressure recovery, increased mixing losses and greater exit flow non-uniformity (or maldistribution) [1, 2]. The diffuser (inlet header) exit forms the inlet to the fin-and-tube heat exchanger and has direct bearing on the flow distribution entering the fin-and-tube heat exchanger [3]. Flow maldistribution created by the geometry of the inlet header is one of the main forms of mechanically induced flow maldistribution [4]. Previous investigations into heat exchanger flow maldistribution effects have mainly focused on heat exchanger thermal performance.

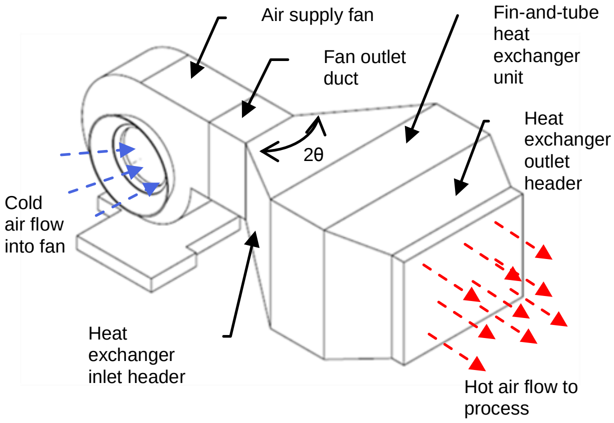

Figure 1 - Typical configuration and components of an industrial air heater unit utilising steam or heating oil in the tube side of the fin-and-tube heat exchanger

Numerical investigations by researchers [5-8] using various compact heat exchanger geometries illustrated effectiveness reductions from air side flow maldistribution of up to 10 percent. Investigations using these numerical methods use the unmixed flow assumption and do not consider the fluid mechanics of the flow maldistribution-heat exchanger interaction. This approach has also been employed on the limited numerical investigations conducted into hydraulic performance effects. Ranganayakulu et al., [9] showed a 248 percent hydraulic resistance increase using a parabolic profile with a maximum normalised velocity magnitude 3.7 times the uniform value. The considerable increase is not unexpected as the numerical method used assumes an unmixed flow distribution where the selected profile interaction with the heat exchanger resistance is ignored. This assumption is also made in industrial compact heat exchanger simulation software such as the Acol+® package from AspenTech [10] where non-uniform flow distributions can be entered. There is no consideration of the flow maldistribution-heat exchanger interaction, leading to an over prediction of the overall hydraulic resistance penalties. The level of over prediction increases with increasing flow maldistribution magnitude.

Experimental investigations overcome the limitations of the numerical unmixed flow assumption. However experimental tests usually only investigate a small number of specific cases. Kitto and Robertson [11] and T'Joen et al. [12] are an example of the commonly investigated small refrigeration/air conditioning type fin-and-tube heat exchangers. A rare example of the experimental investigation into industrial heat exchanger flow maldistribution is the work by Bauver et al., [13] where a 1:12 scale model of a 850 MW steam generator economizer was used to determine details of the industrial flow maldistribution. The resulting modifications improved the boiler efficiency by 0.3 percent (2.55 MW).

Experimental or CFD investigations into fin-and-tube heat exchanger flow distribution or hydraulic resistance variations due to flow maldistribution have not been widespread. The main CFD focus has been on the design of inlet header baffles for compact plate-fin applications [14, 15]. In more relevant applications such as the work by Al-Nasser et al. [16] the fin and tube heat exchanger was over simplified as a one-dimensional radiator thereby considerably reducing the accuracy of the flow maldistribution heat exchanger interaction simulation.

The following CFD modelling aims to provide increased understanding into the flow maldistribution-heat exchanger interaction applicable to industrial air heaters. The sudden expansion inlet condition used to create the flow maldistribution generally replicates the jet flow structures commonly found in industrial air heater inlet headers.

Computational Method

The computational investigation of flow maldistribution-heat exchanger interaction is limited to a two-dimensional expansion with a limited number of tube passes due to computational constraints. A mesh independent solution for the small slice models shown in Figure 2 were used to determine a suitable mesh for the Figure 4 sudden expansion heat exchanger section. As shown in Figure 3 there is very little variation (< 2 %) in the pressure drop with increased grid resolution. The mesh sizing for this domain was then optimised with the purpose of minimising the required mesh for the larger sudden expansion geometry of Figure 4. Mesh resolutions for the two row tube bundle of around 30,000 (similar resolution used in the three row) provided sufficient flow structure resolution and pressure drop accuracy, and were used for the tube bundle section of the Figure 4 modelling domain. The sudden expansion modelling domain provides a two-dimensional expansion allowing analysis of the flow spreading in the plane orientated parallel to the fins. The Z-axis mesh either side of the tube bundle has been rationalised to significantly reduce the total mesh size and allow increased X-Y plane resolution. Analysis of the Figure 2 geometry flow field showed essentially no Z-axis components outside the immediate vicinity of the tube bundle, and only small values inside the tube bundle. The flow structure is thus predominantly two-dimensional. Significant Z-axis mesh resolution is only required inside and in close proximity of the tube bundle to replicate the boundary layer flows across the fins to accurately simulate the pressure drop.

Modelling Domains

Figure 2 shows the two and three row individual circular fin-and-tube geometries used to determine suitable computational settings for use in the sudden expansion geometry of Figure 4. The heat exchanger geometry has a tube outside diameter (X_D) of 0.025 m and a longitudinal spacing (X_L) of 0.061 m. The translation spacing (X_T) is also 0.061 m giving a tube pitch angle of 26.6°. A fin diameter (X_F) of 0.058 m is used for both two and three row cases with the only difference being the extra tube row. The Figure 2 geometries inlet and outlet boundary conditions consist of a uniform un-developed velocity inlet and a pressure outlet exit. The tube and fin walls have been simulated as infinitely thin smooth walls with a no-slip condition. All other modelling domain boundaries have been set as symmetry boundaries. The modelling domain inlet length of $2X_D$ and downstream length of $25X_D$ has been used to ensure a boundary condition independent solution.

Figure 4 shows the general configuration of the sudden expansion geometry and the associated dimensions for the two-row configuration.

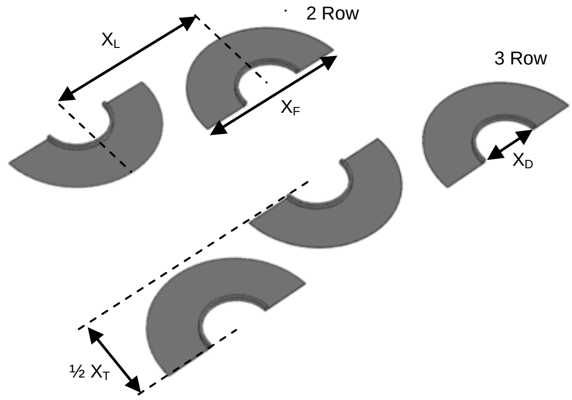

Figure 2 - General configuration and key geometric relations of the two and three row staggered individual circular fin-and-tube heat exchanger geometries

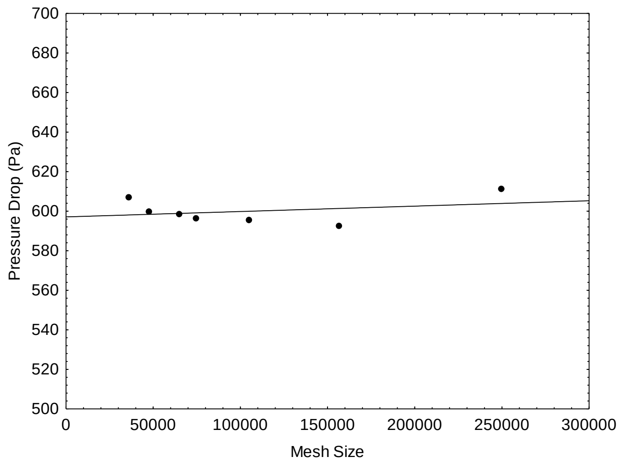

Figure 3 - Grid dependence results for the two-row tube bundle showing a variation of less than 2 percent at a Reynolds number of 27,000 using tube diameter as reference length (Dh of 0.025 m)

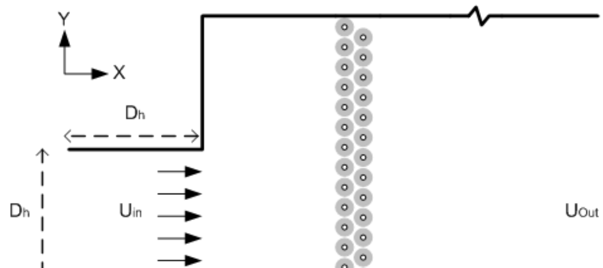

Figure 4 - Showing the two-dimensional sudden expansion geometry and associated dimensions for the two row heat exchanger case (Z-axis thickness of half a fin spacing into the page).

The three row configuration was modelled in a similar manner with the heat exchanger depth increasing due to the extra tube row. An inlet duct hydraulic diameter (Dh) of 0.55 m was used and the modelling geometry dimensions have been constructed in relation to this dimension. While the sudden expansion geometry is three-dimensional, the Z-axis thickness is only half a fin spacing of 0.00175 m. The two-dimensional area ratio three expansion created by the sudden expansion geometry allows the X-Y plane interaction between the flow maldistribution and the heat exchanger to be modelled on an industrially relevant scale. The sudden expansion inlet height (D_h) is 22 times the tube diameter to ensure a sufficient jet size in relation to the heat exchanger geometry. Flow maldistribution jet sizes of a similar magnitude to the heat exchanger tube diameter are not relevant to industrial air heater flow maldistribution cases.

The sudden expansion modelling domains utilise an undeveloped velocity inlet to the sudden expansion and a pressure outlet downstream of the heat exchanger. The modelling domain top and bottom walls are set as smooth with a no-slip condition. The front and back face of the modelling domain is set as a symmetry boundary condition. Overall the two row sudden expansion model has 2.3 million mesh cells and the three row model uses 2.8 million cells.

Computational Settings

The current modelling only focuses on flow distribution and hydraulic resistance and uses isothermal conditions ignoring the small increase in hydraulic resistance associated with heating air. The CFD has been carried out using Fluent 6.3 with geometry and mesh generation through Gambit 2.4. The tube bundle slice models of Figure 2 were solved using a segregated steady state solver. The Figure 4 sudden expansion section models were solved using a segregated transient solver due to the time dependent vortex shedding. Models were solved using second order discretization for all components with pressure velocity coupling using SIMPLEC. Turbulence modelling was carried out using the standard k-ω turbulence model of Wilcox [17]. Convergence of the steady state models was determined using an inlet static pressure monitor in conjunction with residuals. The transient sudden expansion models used time averaged inlet static pressure monitors in conjunction with residual convergence to 1x10-4. Mass flux balance at these residuals was very good for the converged solution. Air at constant density of 1.2 kg/m³ was used as the working fluid. Solution time for each sudden expansion model ranged between three and six weeks depending on the sudden expansion inlet velocity using six Core 2 Due 3.0GHz processors connected in parallel.

Results

The two and three row pass arrangements used in the current analysis represent the initial passes of industrial air heaters exposed to incoming air flow. The initial passes are commonly condensate or recuperator sections. Commonly air heater heat exchangers consist of a series of one, two or three row passes which are spaced to allow for header connections on the outside of the heat exchanger between individual sections. The current simulations provide an insight into the likely interaction between jet flow structures in the inlet header and the initial passes of an air heater fin-and-tube heat exchanger.

Figure 5 shows the computational isothermal hydraulic resistance of the two and three row heat exchanger slice sections shown in Figure 2 with uniform inlet flow distribution. The extra row in the three row tube bundle results in a higher overall hydraulic resistance.

An example of the sudden expansion geometry results are shown in the velocity magnitude contour plots for the 20 m/s inlet jet cases for the two and three row heat exchangers in Figure 6. The flow distribution for the 10 m/s and 20 m/s inlet jet velocity is very similar and as a result only one inlet velocity is presented.

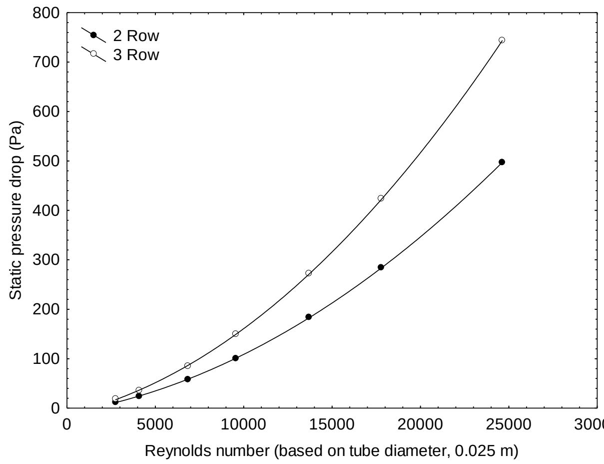

Figure 5 - Computational isothermal pressure drop curves for the two and three row individual circular fin-and-tube geometries of Figure 2 with uniform inlet velocity distribution.

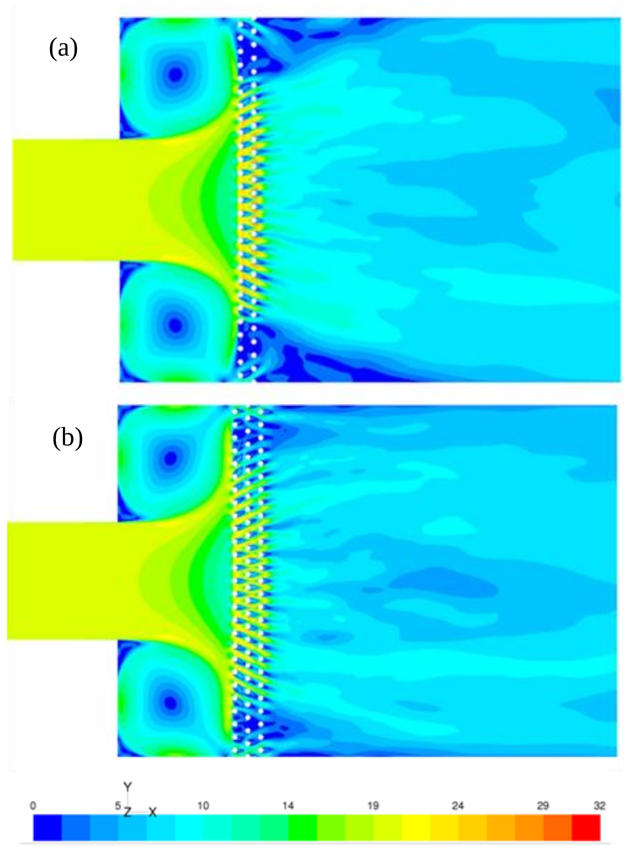

Figure 6 - Time averaged velocity magnitude contours for the sudden expansion geometries for the 20 m/s inlet jet (a) two row tube bundle (b) three-row tube bundle

The general difference between the two and three row cases is the amount of flow maldistribution dispersion occurring due to the heat exchanger resistance. While both heat exchanger configurations spread the inlet flow maldistribution, areas of low flow are still present. Regions of low flow are largest in the two-row heat exchanger where close to one third of the heat exchanger receives comparatively little flow. The greater uniformity of the flow downstream of the fin-and-tube tube bundle highlights that the downstream fin-and-tube bundles are likely to be less effected by the inlet flow maldistribution.

Inlet flow maldistribution affects on the fin-and-tube bundle pressure drop are shown in Table 1. Flow maldistribution pressure drop values also consider the dynamic pressure changes associated with the expanding sudden expansion jet. Through consideration of the total pressure change the uniform and flow maldistribution system pressure changes can be suitably compared. In all cases, there is a significant increase in system pressure loss in the flow maldistribution cases. The system pressure drop for the three-row case is greater than for the two-row cases. The two-row pressure drop increase over the uniform benchmark is on average 56 percent higher than that of the three-row case. The greater pressure drop penalty is due to the lower flow maldistribution spreading as shown in Figure 6.

Static pressure changes in the immediate vicinity (from front leading fin tip to back trailing fin tip) of the fin-and-tube bundle are similar for both two and three-row cases. Due to the higher velocities through the two-row case, a greater proportion of the static pressure change is flow acceleration related. The downstream recovery of the greater two-row flow acceleration static pressure loss results in a higher system static pressure drop for the three-row case.

Table 1 - System pressure drop increase for flow maldistribution cases

Tube rows	Inlet jet velocity (m/s)	System total pressure drop (Pa)		Increase over uniform case
		Uniform flow	Inlet flow maldistribution	
2	10	29.2	69.2	137 %
3	10	43.3	80.52	86 %
2	20	93.0	259.7	179 %
3	20	138.5	302.7	118 %

Conclusions

Transient CFD modelling of flow maldistribution-heat exchanger interaction highlights the importance of considering both flow maldistribution magnitude and heat exchanger resistance. For a specific degree of flow maldistribution and specific heat exchanger geometry a greater number of heat exchanger passes (higher resistance) will result in a more uniform heat exchanger flow distribution, and lower flow maldistribution related pressure drop penalty. The extra heat exchanger passes will also provide the benefit of greater heat transfer surface area, while also reducing the size of the dead zones found in the heat exchanger. Higher resistance heat exchanger sections will result in less potential flow maldistribution related performance reductions. One and two tube row fin-and-tube bundle recuperator sections at the front of air heaters are likely to experience the most severe effects of any inlet header flow maldistribution due to their lower flow spreading ability.

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